## REMARKS

Please find attached copies of pages from reference book: John C. Dixon, "Tires, Suspension and Handling", 2nd ed., Society of Automotive Engineers, Inc., Warrendale, Pa.

Applicant's comments with respect to this submission and further explanation of the claimed feature, "response frequency of change in behavior of the vehicle," are below:

In the present application, "response frequency," relates to a vehicle's natural frequency of vibration (its *response frequency*). In an exemplary embodiment of the present invention, a vehicle body and tires are connected through suspension springs and, since the inertial mass of the vehicle is large, there is a delay, based on the response frequency of the vehicle, in the vehicle's response to an input to the tires. Here, the response frequency of the vehicle can depend on the mass of the vehicle as well as the particulars of the suspension, etc., and is typically around 1 Hz or so.

In claim 1, for example, it is recited that the vibration that is applied to a tire is a micro-vibration having a higher frequency than the above-described response frequency (the vehicle's natural frequency of vibration). That is, even though a particular vehicle's response frequency may change based on changing/different factors, the claimed micro-vibration that is applied has a higher frequency than said response frequency. Applicants submit that one skilled in the art would understand the claimed relationship between the frequency of the micro-vibration and the response frequency of change in behavior of the vehicle.

In view of the above, the attached reference materials, and previously submitted arguments, reconsideration and allowance of this application are now believed to be in order, and such actions are hereby solicited. If any points remain in issue which the Examiner feels may be

best resolved through a personal or telephone interview, the Examiner is kindly requested to contact the undersigned at the telephone number listed below.

The USPTO is directed and authorized to charge all required fees, except for the Issue Fee and the Publication Fee, to Deposit Account No. 19-4880. Please also credit any overpayments to said Deposit Account.

Respectfully submitted,

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# Tires, Suspension and Handling

## Second Edition

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This suffices to find $A$ and $B$, as required.

For the case of a neutral-steer vehicle, the standard solution is

$$r = A\,e^{D_1 t} + Bt\,e^{D_1 t} + r_1$$

At $t = 0$

$$r_0 = 0 = A + r_1$$

Differentiating:

$$\dot{r} = AD_1\,e^{D_1 t} + B\,e^{D_1 t} + BD_1 t\,e^{D_1 t}$$

At $t = 0$

$$\dot{r}_0 = \frac{2aC_{\alpha f}\delta}{I} = AD_1 + B$$

This suffices to find $A$ and $B$ for the neutral-steer case.

If it is required to find the attitude angle rather than the yaw speed, then $r$ must be eliminated from the simultaneous differential equations of motion rather than $\beta$. To study the path of the vehicle, for example to find the path of G, the Earth-fixed axes formulation must be used. In all cases the method of solution is as above.

For the three-degrees-of-freedom model, the analysis is more complex but gives very similar results.

## 7.12 Oscillatory Steer Response

The "steady-state" response to an oscillatory (sinusoidal) steer input is not of reat practical significance, since this is unlikely to arise in normal operation on ie road. However, a sinusoidal steer input is of some theoretical interest for a equency domain analysis, and can be investigated experimentally.

The three-degrees-of-freedom analysis for this input was first performed by egel (1955a,b). Figures 7.12.1 to 7.12.3 show how the yaw speed amplitude, ll speed amplitude and lateral acceleration amplitude varied with input freency. Modern cars and tires tend to give flatter responses up to rather higher equencies. Modern coaches can produce results much as in these figures. For equencies below 3 Hz, provided that the speed is not very low, the tire dynamic

characteristic (lag between force and angle) can be neglected. As would be expected, the yaw velocity amplitude (Figure 7.12.1), diminishes with frequency, beginning with the steady-state response value. At high frequency it is inertia-limited, so there is only a small yaw amplitude. Neglecting the small response, then evidently the yaw moment tends to $M_z = 2aC_{\alpha f}\delta$, where $\delta$ is the applied steer angle amplitude. The yaw acceleration amplitude tends to $A_y = 2aC_{\alpha f}\delta/I$ and the yaw velocity amplitude to $r = 2aC_{\alpha f}\delta/I\omega_f$, where $\omega_f$ is the forcing radian frequency of the steering motion. The phase of the response varies from in-phase at low frequency to 90° lag at high frequency.

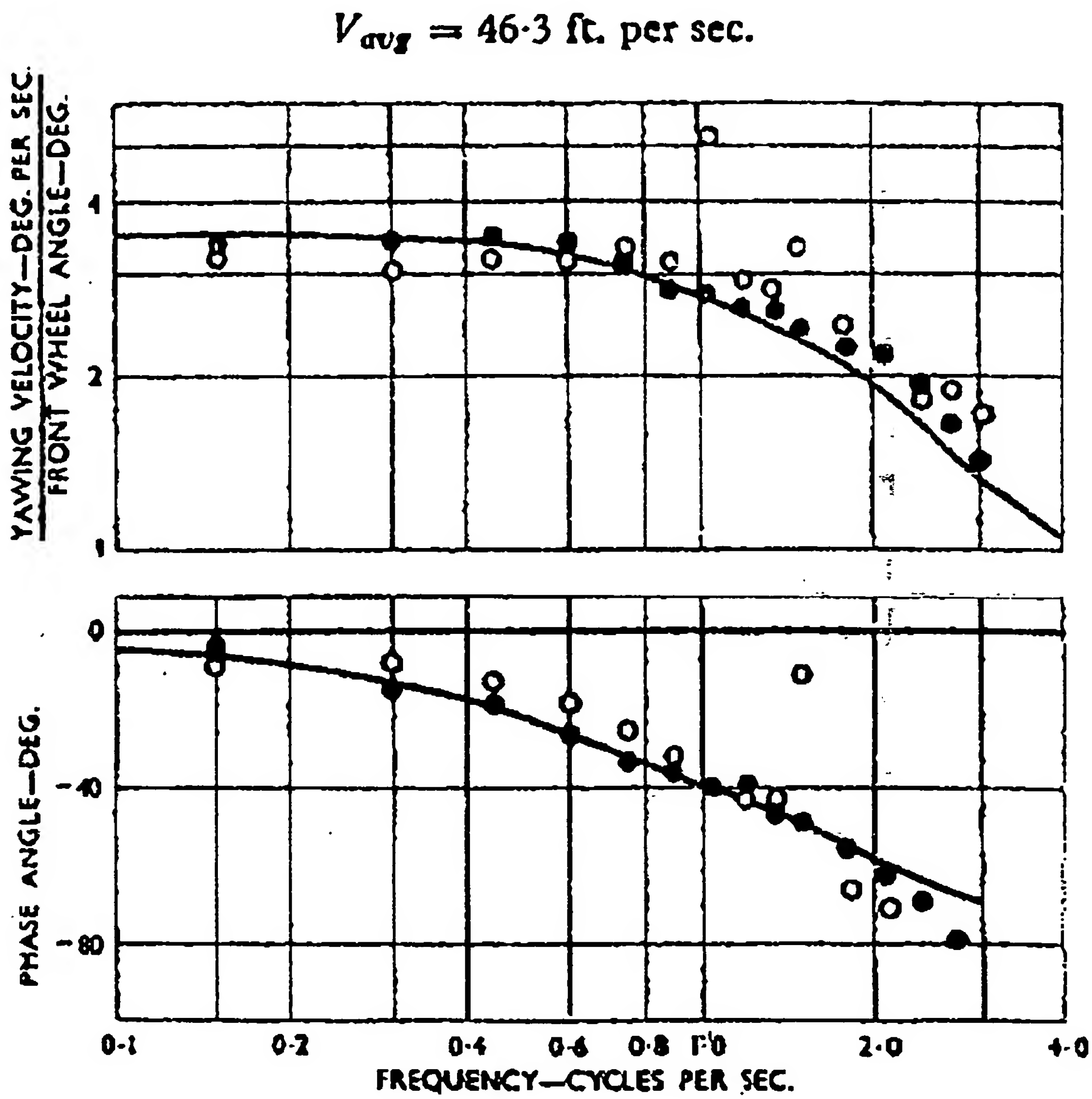
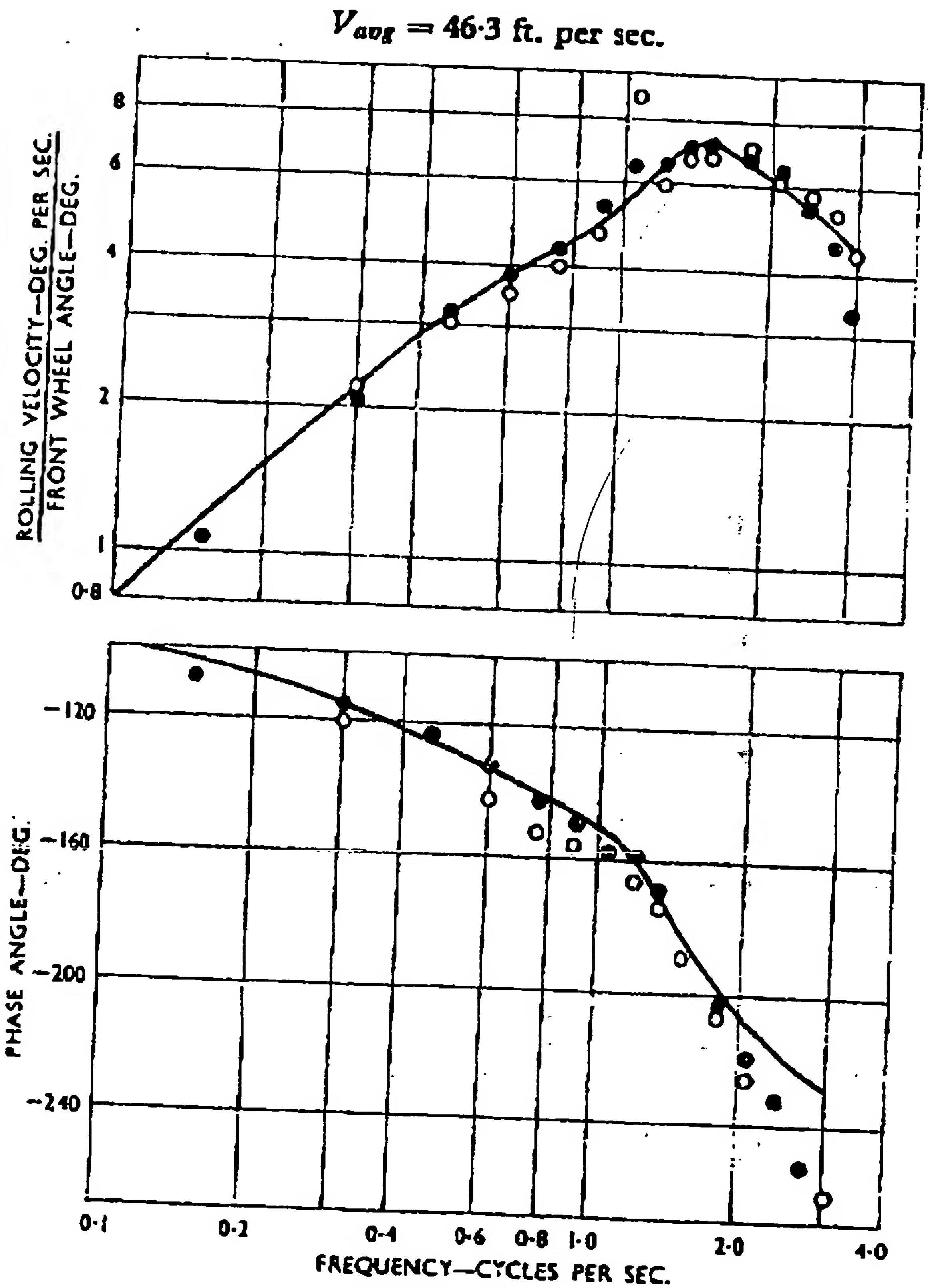



*Figure 7.12.1. Yaw speed amplitude and phase: against steer frequency. (Reproduced by permission of the Council of the Institution of Mechanical Engineers, from Segel, L., "Theoretical Prediction and Experimental Substantiation of the Response of the Automobile to Steering Control,"* Proc. I.Mech.E., *1957.)*




*ure 7.12.2. **Roll speed amplitude and phase: against steer frequency.** (Repro-
d by permission of the Council of the Institution of Mechanical Engineers, from
, L., "Theoretical Prediction and Experimental Substantiation of the Response
of the Automobile to Steering Control,"* Proc. I.Mech.E., *1957.)*

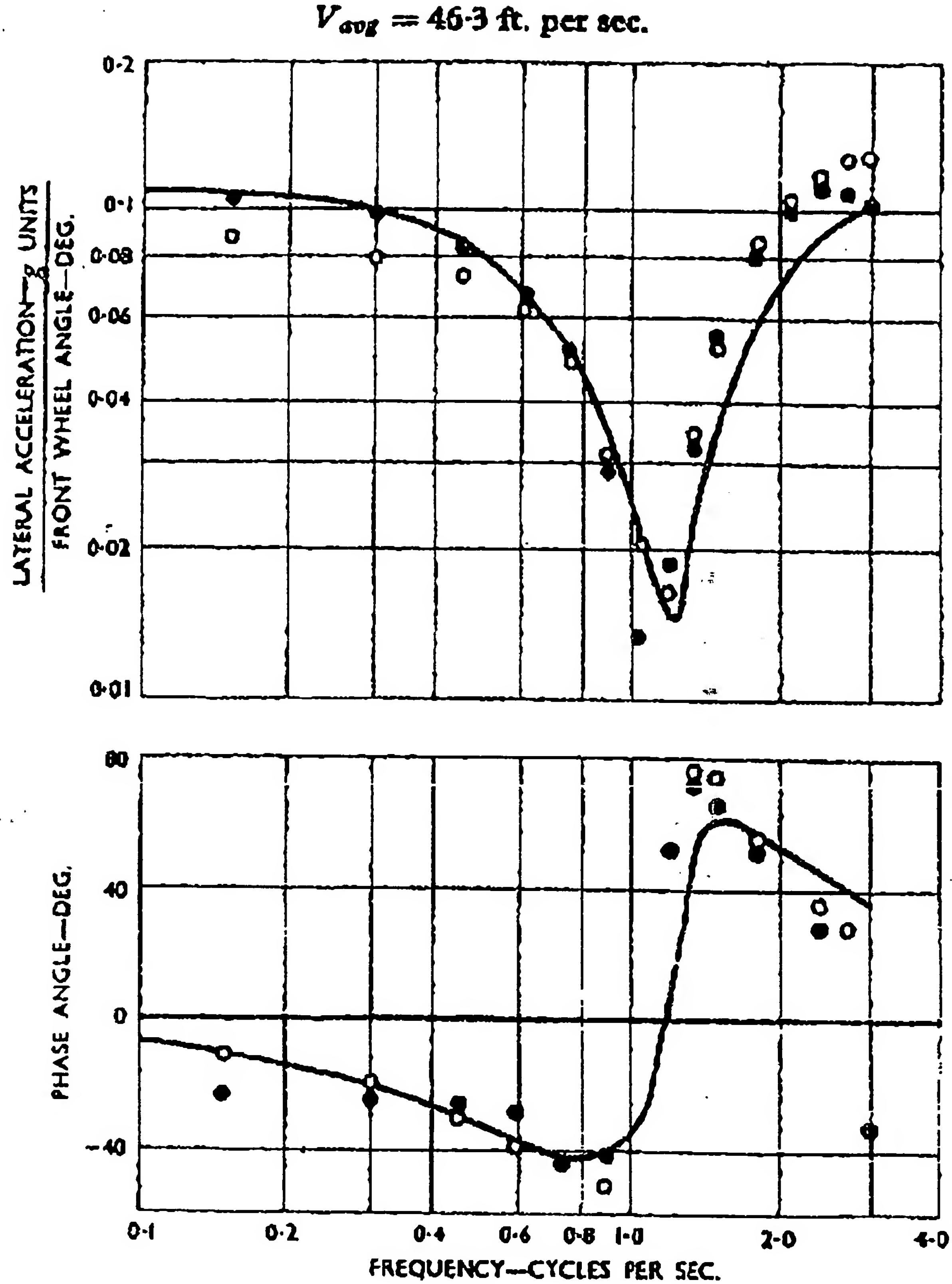



*Figure 7.12.3. Lateral acceleration amplitude and phase: against steer frequency. (Reproduced by permission of the Council of the Institution of Mechanical Engineers, from Segel, L., "Theoretical Prediction and Experimental Substantiation of the Response of the Automobile to Steering Control,"* Proc. I.Mech.E., *1957.)*

The roll velocity response (Figure 7.12.2), begins at zero at low frequency, increases to a peak at around the roll natural frequency, and then declines. The phase lag goes from 90°, through 180° around resonant frequency, to 270° lag at high frequency.

The shape of the lateral acceleration response, Figure 7.12.3, is somewhat unexpected, being rather like an inverted resonance curve. The agreement between theory and practice is, therefore, particularly gratifying. At low frequency the value equals the steady-state response. At high frequency there is little yaw response, so for the 2-dof model the total amplitude side force tends to approximately $F_y = 2C_{\alpha f}\delta$ and the lateral acceleration amplitude to $A_y = 2C_{\alpha f}\delta/m$. In between, there is a minimum. The reason for this shape of curve is that it is the sum of three terms, two decreasing with frequency, and one, the linear lateral acceleration, increasing with frequency. The phase of the response begins at zero, goes to about 45° lead through the resonant frequency, and then declines to zero again.

If the steering forcing frequency is increased beyond about 3 Hz then the tire dynamics will become significant, and there will be a further reduction in response and an extra phase lag.

Instead of using a sinusoidal steering motion of various given frequencies, an alternative frequency domain test is to use a random steer input. The vehicle motion is then the product of the random input spectrum and the vehicle transfer function, so by correlating the output motion with the input steering, the transfer function may be deduced. This is particularly easy if the input is white noise, i.e., having a uniform spectral distribution, at least over the frequency range of interest.

## .13 Power Steer

Previous sections have dealt with the response to steering control, but there may also be steering response to the accelerator or brake, known as power steer and brake steer, respectively. The former must obviously be distinguished from "power steering," i.e., power-assisted steering. Such effects are generally undesirable and are designed out if possible (Section 5.13). The effects may be considered to arise in two ways. The first is by effects that occur on the vehicle considered as a rigid body and include differences in tractive force from side to side, for example because of a limited-slip differential or because of tractive force effects on the tire characteristics. The second is because of vehicle internal compliances.

The dynamic effects are similar to those in the steady state, but may be much greater because the tractive forces are greater. In particular, "lift-off tuck-in" may be very noticeable – really this is the disappearance of power understeer, although nowadays it is known how to control this or to eliminate it if required.

Where a large amount of power is applied then the driven end of the vehicle will inevitably have its cornering ability reduced because of the tire characteristics, giving oversteer for rear drive and understeer for front drive. For more moderate power applications, many details matter and one cannot really generalize. The effect of internal compliance is also very complex and will not be dealt with here, but see Section 5.13.

The effect of a non-free differential on the vehicle as a rigid body may be analyzed in the following way. The tractive force above that required for steady state is $F = mA_X$. Consider a rear-drive vehicle with a tractive force lateral transfer factor $e_T$:

$$F_i = \frac{1}{2} F(1 - e_T)$$

$$F_o = \frac{1}{2} F(1 + e_T)$$

With $e_T = 0$ the traction is equally distributed as on a free differential; with $e_T = 1$ all the traction is on the outer wheel. The plan-view moment of the tractive force is

$$M = \frac{1}{2} F e_T T$$

To avoid changing the steady-state cornering radius there must be counteracting side forces at the axles of

$$\frac{M}{L} = \frac{F e_T T}{2L}$$

The corresponding changes of slip angle are

$$\Delta\alpha_f = -\frac{F e_T T}{4 L C_{\alpha f}}$$

$$\Delta\alpha_r = \frac{F e_T T}{4 L C_{\alpha r}}$$